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Preshaping Command Inputs to Reduce System Vibration

Neil C. Singer
Warren P. Seering

Abstract: A method is presented for generating shaped command inputs which significantly reduce or eliminate endpoint vibration. Desired system inputs are altered so that the system completes the requested move without residual vibration. A short move time penalty is incurred (on the order of one period of the first mode of vibration). The preshaping technique is robust under system parameter uncertainty and may be applied to both open and closed loop systems. The Draper Laboratory's Space Shuttle Remote Manipulator System simulator (DRS) is used to evaluate the method. Results show a factor of 25 reduction in endpoint residual vibration for typical moves of the DRS.

Note: The methods described in this paper are patent pending. Commercial use of these methods requires permission from the Massachusetts Institute of Technology

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1 Introduction

Input command shaping and closed-loop feedback for vibration control are two distinct approaches toward vibration reduction of flexible systems. Many researchers have examined closed-loop feedback techniques for reducing endpoint vibration, for example [9][26][19]. These techniques differ from input shaping in that they use measurements of the system's states to reduce vibration. Command shaping involves altering the shape of either actuator commands or setpoints so that system oscillations are reduced [16] [23]. This technique is often dismissed because it is mistakenly considered to be useful only for open loop systems. However, if the input shaping accounts for the dynamic characteristics of the closed loop plant, then shaped input commands can be given to the closed loop plant as well. Thus, any of the preshaping techniques may be readily used as a closed loop technique [16] [23].

The earliest form of command preshaping was the use of high-speed cam profiles as motion templates. These input shapes were generated so as to be continuous throughout one cycle (ie. the cycloidal cam profile). Their smoothness (continuous derivatives) reduces unwanted dynamics by not putting high frequency inputs into the system [20]; however, these profiles have limited success.

Another early form of setpoint shaping was the use of posicast control by O.J.M. Smith [23]. This technique involves breaking a step of a certain magnitude into two smaller steps, one of which is delayed in time. This results in a response with a reduced settling time. In effect, superposition of the responses leads to vibration cancellation. However, this is not generally used because of problems with robustness. The system that is to be commanded must have only one resonance, be known exactly, and be very linear for this technique to work.

Optimal control approaches have been used to generate input profiles for commanding vibratory systems. Junkins, Turner, Chun, and Juang have made considerable progress toward practical solutions of the optimal control formulation for flexible systems[10][11][5]. Typically, a penalty function is selected (for example integral squared error plus some control penalty). The resulting "optimal" trajectory is obtained in the form of the solution to the system equations (a model). This input is then given to the system.

Farrenkopf [6] and Swigert [24] demonstrated that velocity and torque shaping can be implemented on systems which modally decompose into second order harmonic oscillators. They showed that inputs in the form of the solutions for the decoupled modes can be added so as not to excite vibration while moving the system. Their technique solves for parameters in a template function, therefore, inputs are limited to the form of the template. These parameters that define the control input are obtained by minimizing some cost function using an optimal formulation. The drawback of this approach is that the inputs are difficult to compute and they must be calculated for each move of the system.

Gupta [8], and Junkins and Turner [10] also included some frequency shaping terms in the optimal formulation. The derivative of the control input is included in the penalty function so that, as with cam profiles, the resulting functions are smooth.

Several papers also address the closed loop "optimal" feedback gains which are used in conjunction with the "optimal" open-loop input. [10][11][5]

There are four drawbacks to these "optimal" approaches. First, computation is difficult. Each motion of the system requires recomputation of the control. Though the papers cited above have made major advances toward simplifying this step, it continues to be extremely difficult or impossible to solve for complex systems.

Second, the penalty function does not explicitly include a direct measure of the unwanted dynamics (often vibration). Tracking error is used in the penalty function, therefore, all forms of error are essentially lumped together – the issue of unwanted dynamics is not addressed directly. One side effect is that these approaches penalize residual vibration but allow the system to vibrate during the move. This leads to a lack of robustness under system uncertainties. In addition, vibration during a move may be undesirable.

Third, the solutions are limited to the domain of continuous functions. This is an arbitrary constraint which enables the solution of the problem. Fourth, the value of optimal input strategies depend on move time. Different moves will have different vibration excitation levels.

Another technique is based on the concept of the computed torque approach. The system is first modeled in detail. This model is then inverted — the desired output trajectory is specified and the required input needed to generate that trajectory is computed. For linear systems, this might involve dividing the frequency spectrum of the trajectory by the transfer function of the system, thus obtaining the frequency spectrum of the input. For nonlinear systems this technique involves inverting the equations for the model. [1]

Techniques that invert the plant have four problems. First, a trajectory must be selected. If the trajectory is impossible to follow, the plant inversion fails to give a usable result. Often a poor trajectory is selected to guarantee that the system can follow it, thus defeating the purpose of the input [3]. Second, a detailed model of the system is required. This is a difficult step for machines which are not simple. Third, the plant inversion is not robust to variations in the system parameters because no robustness criterion has been included in the calculation. Fourth, this technique results in large move time penalties because the plant inversion process results in an acausal input (an input which exists before zero time). In order to use this input, it must be shifted in time thus increasing the move time.

Another approach to command shaping is the work of Meckl and Seering [12] [13] [14] [15] [16][17]. They investigated several forms of feedforward command shaping. One approach they examined is the construction of input functions from either damped sinusoids or versine functions. This approach involves adding up harmonics

of one of these template functions. If all harmonics were included, the input would be a time optimal rectangular (bang-bang) input function. The harmonics that have significant spectral energy at the natural frequencies of the system are discarded. The resulting input which is given to the system approaches the rectangular shape, but does not significantly excite the resonances. This technique essentially constructs an input function prior to a move. The approach presented in this paper does not require continuous functions and the processing can be performed in real-time.

Aspinwall [2] proposed a similar approach which involves creating input functions by adding harmonics of a sine series. The coefficients of the series are chosen to minimize the frequency content of the input over a band of frequencies. Unlike Meckl, the coefficients were not selected to make the sine series approach a rectangular function, therefore, a large time penalty was incurred.

Wang, Hsia, and Wiederrich [25] proposed yet another approach for creating a command input that moves a flexible system while reducing the residual vibrations. They modeled the system in software and designed a PID controller for the plant that gave a desired response. They then examined the actual input that the controller gave to the software plant and used this for the real system. Next, they refined this input (the reference) with an iteration scheme that adds the error signal to the reference in order to get better tracking of the trajectory. This technique requires accurate modeling of the system and is not robust to parameter uncertainty. In addition, the method assumes that a good response can be achieved with a PID controller, In fact, systems with flexibility can not, in general, be given sufficient damping and a reasonable response time by adding a PID controller.

Often, a notch filter is proposed for input signal conditioning. This approach gives poor results for several reasons. First, a causal (real time) filter distorts the phase of the resulting signal. This effect is aggravated by lengthening the filter sequence of digital filters or by increasing the order of analog or recursive filters. Therefore, efforts to improve the frequency characteristics of a filter result in increased phase distortion. Also, penalties, such as filter ringing or long move times often result.

Singer and Seering [21] investigated an alternative approach of shaping a time optimal input by acausally filtering out the frequency components near the resonances. This has an advantage over notch filtering in that phase distortion and ringing no longer pose a problem. The drawbacks of this approach [21] are the tradeoffs that must be made between fidelity in frequency and reduction of the move time.

2 Shaping Inputs

Most researchers have examined the transient vibration of manipulators in terms of frequency content of the system inputs and outputs. This approach inherently assumes that the system inputs are not actually transient, but are one cycle of a

repeating waveform. The approach taken in this paper is fourfold: first, the transient residual vibration amplitude of a system will be directly expressed as a function of its transient input. Second, the input will be specified so that the system's natural tendency to vibrate is used to cancel residual vibration. Third, the input will be modified to include robustness to uncertainties. Fourth, the case of arbitrary system inputs will be examined.

2.1 Generating a Vibration-Free Output

The derivation of the new technique will be based on linear system theory. The results obtained will then be demonstrated on a more complicated system. The first step toward generating a system input which results in a vibration-free system output is to specify the system response to an impulse input. An uncoupled, linear, vibratory system of any order can be specified as a cascaded set of second-order poles with the decaying sinusoidal response [4]:

$$y(t) = \left[A\frac{\omega_0}{\sqrt{1.0-\zeta^2}}e^{-\zeta\omega_0(t-t_0)}\right]\sin\omega_0\sqrt{1.0-\zeta^2}(t-t_0) \quad , \tag{1}$$

where A is the amplitude of the impulse, ω_0 is the undamped natural frequency of the plant, ζ is the damping ratio of the plant, t is time, and t_0 is the time of the impulse input. The impulse is usually a torque or velocity command to an actuator. Equation 1 specifies the acceleration or velocity response, $y(t)$, at some point of interest in the system.

In this section, only one mode is assumed (the general case is treated in section 2.3). Figure 1 demonstrates that two impulse responses can be superposed so that the system moves forward without vibration after the input has ended. In this case, the input consists of two impulses; the "end" or duration of the input is the time of the last (second) impulse. The same result can be obtained mathematically by adding two impulse responses [each described by (1)], and expressing the result for all times greater than the duration of the input. Using the trigonometric relation (from [7]):

$$B_1\sin(\alpha t+\phi_1)+B_2\sin(\alpha t+\phi_2) = A_{\mathrm{amp}}\sin(\alpha t+\psi) \quad , \tag{2}$$

where

$$\begin{aligned} A_{\mathrm{amp}} &= \sqrt{(B_1\cos\phi_1+B_2\cos\phi_2)^2+(B_1\sin\phi_1+B_2\sin\phi_2)^2} \\ \psi &= \tan^{-1}\left(\frac{B_1\cos\phi_1+B_2\cos\phi_2}{B_1\sin\phi_1+B_2\sin\phi_2}\right) \quad . \end{aligned}$$

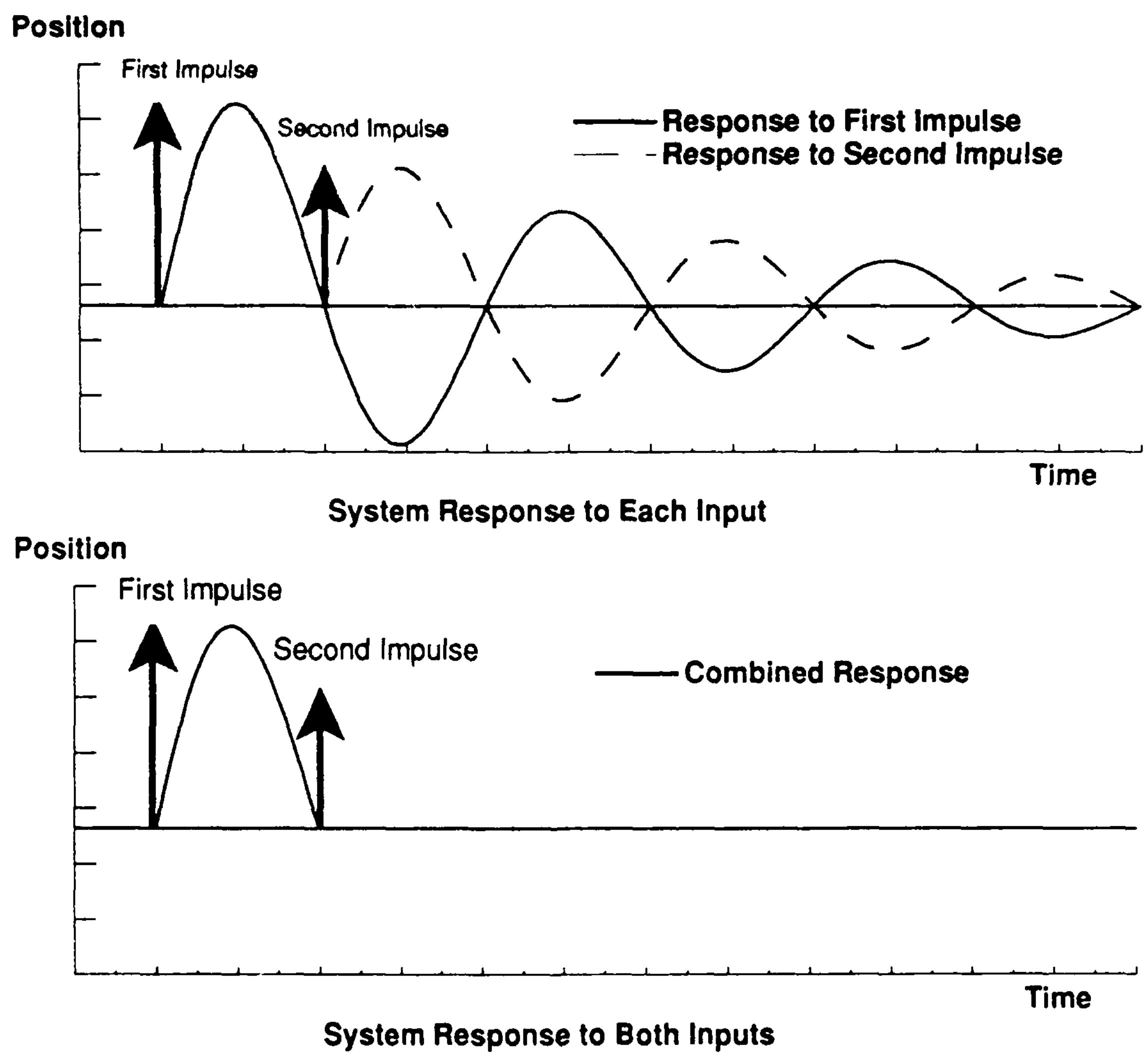


Figure 1: The two impulse responses shown add to form an output that shows a net positive motion with no vibration after the input has ended at the time of the second impulse.

The amplitude of vibration for a multi-impulse input is given by:

$$A_{\text{amp}} = \sqrt{\left(\sum_{j=1}^{N} B_j \cos\phi_j\right)^2 + \left(\sum_{j=1}^{N} B_j \sin\phi_j\right)^2} \tag{3}$$

$$\phi_j = \omega_0\sqrt{(1-\zeta^2)}t_j \quad .$$

The B_j are the coefficients of the sine term in (1) for each of the N impulse inputs, and the t_j are the times at which the impulses occur. Elimination of vibration after the input has ended requires that the expression for A_{amp} equal zero at the time at which the input ends, t_N. This is true if both squared terms in 3 are independently zero, yielding:

$$B_1\cos\phi_1 + B_2\cos\phi_2 + \cdots + B_N\cos\phi_N = 0 \tag{4}$$

$$B_1\sin\phi_1 + B_2\sin\phi_2 + \cdots + B_N\sin\phi_N = 0 \quad , \tag{5}$$

with

$$B_j = \frac{A_j\omega_0}{\sqrt{(1-\zeta^2)}}e^{-\zeta\omega_0(t_N-t_j)} \quad ,$$

where A_j is the amplitude of the Nth impulse, t_j is the time of the Nth impulse, and t_N is the time at which the sequence ends (time of the last impulse). Equations 4 and 5 can be simplified further yeilding:

$$\sum_{j=1}^{N} A_j e^{-\zeta\omega(t_N-t_j)}\sin\left(t_j\omega\sqrt{1-\zeta^2}\right) = 0$$

$$\sum_{j=1}^{N} A_j e^{-\zeta\omega(t_N-t_j)}\cos\left(t_j\omega\sqrt{1-\zeta^2}\right) = 0 \tag{6}$$

If the input is chosen so that there are N impulses, N terms must be included in equation 6.

For the two-impulse case, only the first two terms exist in (6). By selecting 0 for the time of the first impulse (t_1), and 1 for its amplitude (A_1), two equations (6) with two unknowns (A_2 and t_2) result. A_2 scales linearly for other values of A_1. The solution of these two equations produces the input sequence shown in figure 2. The detailed derivation of this result is long and can be found in Singer [22]

2.2 Robustness

2.2.1 Robustness to Errors in Natural Frequency

The two-impulse input, however, cancels vibration only if the system natural frequency and damping ratio are exact. In order to quantify the residual vibration

level for a system, a vibration-error expression must be defined, here as the maximum amplitude of the residual vibration during a move as a percentage of the amplitude of the rigid body motion. This definition is expressed mathematically with equation 3 divided by the sum of all the A_j. Figure 3 shows a plot of the vibration error as a function of the system's actual natural frequency. The input was designed for a system with a natural frequency of ω_0. Acceptable response is defined as less than 5% residual vibration [18]. Figure 3 shows that the two-impulse input is robust for a frequency variation of less than $\approx \pm 5\%$.

In order to increase the robustness of the input under variations of the system natural frequency, a new constraint may be added. The derivatives of (6) with respect to *frequency* (ω_0) can be set equal to zero — the mathematical equivalent of setting a goal of small changes in vibration error for changes in natural frequency. The two equations for these derivatives,

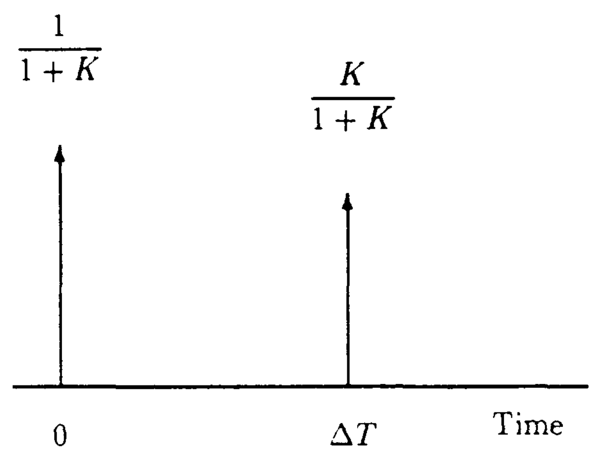
$$\begin{aligned}
\sum_{j=1}^{N} A_j t_j e^{-\zeta\omega(t_N - t_j)} \sin\left(t_j \omega \sqrt{1-\zeta^2}\right) &= 0 \\
\sum_{j=1}^{N} A_j t_j e^{-\zeta\omega(t_N - t_j)} \cos\left(t_j \omega \sqrt{1-\zeta^2}\right) &= 0 \qquad . \qquad (7)
\end{aligned}$$

are added to the system; therefore, two more unknowns must be added by increasing the input from two to three impulses (added unknowns: A_3 and t_3). The details of the derivation of this result are given in Singer [22]. The corresponding input and vibration error curves that result from solving the four equations are shown in figures 4 and 5. In this case, the input is robust for system frequency variations of $\approx \pm 20\%$.

The process of adding robustness can be further extended to include the second derivatives of (6) with respect to ω_0. the general form of the qth derivative of equation 6 with respect to ω is given by:

$$\begin{aligned}
\sum_{j=1}^{N} A_j (t_j)^q e^{-\zeta\omega(t_N - t_j)} \sin\left(t_j \omega \sqrt{1-\zeta^2}\right) &= 0 \\
\sum_{j=1}^{N} A_j (t_j)^q e^{-\zeta\omega(t_N - t_j)} \cos\left(t_j \omega \sqrt{1-\zeta^2}\right) &= 0 \qquad (8)
\end{aligned}$$

Setting the second derivatives (equation 8 when $q = 2$) to 0 requires that the vibration error be flat around the intended natural frequency. Two more constraint equations are added, therefore, the impulse sequence is increased by one to a total of four impulses. The corresponding input and vibration error curves are shown in figures 6 and 7. In this case, the input is robust for system frequency variations of $\approx -30\% + 40\%$.



$$
\begin{aligned}
K &= e^{-\frac{\zeta\pi}{\sqrt{1-\zeta^2}}} \\
\Delta T &= \frac{\pi}{\omega_0\sqrt{1-\zeta^2}}
\end{aligned}
$$

Figure 2: Two-impulse input — designed to have a vibration-error expression which is zero at the expected system natural frequency, ω_0. ζ is the expected damping ratio. Note that K happens to be the expression for the step response overshoot of a 2-pole linear system with no numerator dynamics, and ΔT is the time of the first overshoot.

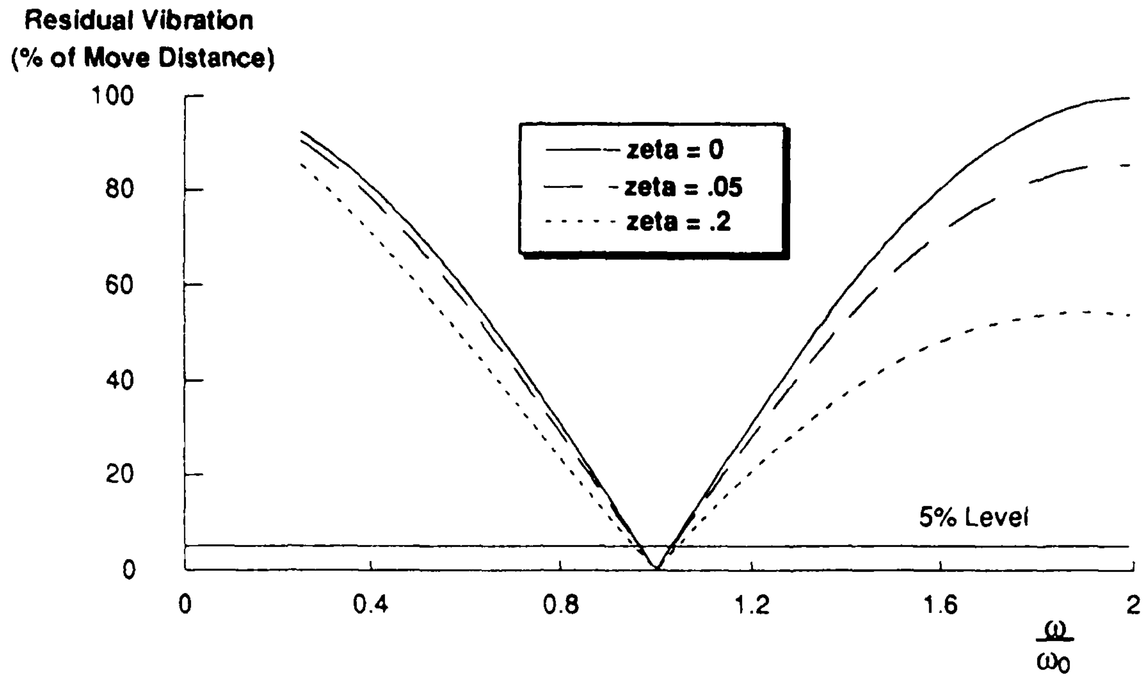


Figure 3: Vibration error vs. system natural frequency for three systems with different values of damping ratio excited by the two-impulse sequence in figure 2.

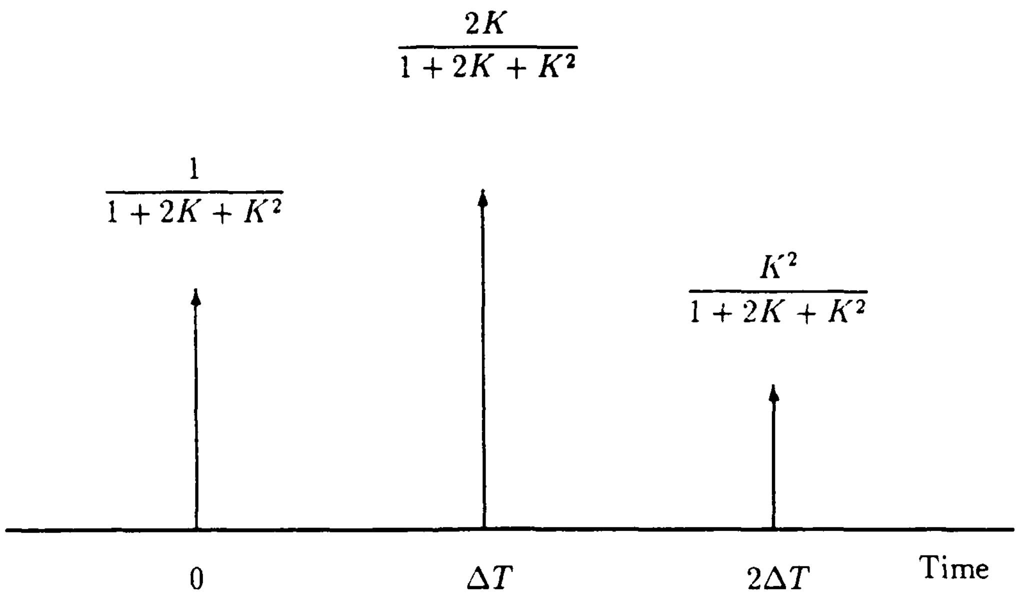
2.2.2 Robustness to Errors in Damping

In order for these system inputs to be insensitive to system parameter variation, uncertainty in damping ratio must also be considered. As with respect to natural frequency in the previous section, the derivative of the amplitude of vibration with respect to damping ratio (ζ) can be computed. It can be shown [22] that the same expressions that guarantee zero derivatives with respect to frequency also guarantee zero derivatives with respect to damping ratio. Therefore, robustness to errors in damping has already been achieved by the addition of robustness to errors in frequency. Figure 8 shows the vibration-error expression for the same three sequences as were generated in 2.2.1. Note that extremely large variations in damping are tolerated.

2.3 Including Higher Modes

The previous sections have assumed only one vibrational mode present in the system. However, the impulse sequence can easily be generalized to handle higher modes. If an impulse or pulse sequence is designed for each of the first two modes of a system independently, they can be convolved to form a sequence which moves a two-mode system without vibration. Figure 9 demonstrates this on two sequences.

The length of the resulting sequence is the sum of the lengths of the individual



$$
\begin{aligned}
K &= e^{-\frac{\zeta\pi}{\sqrt{1-\zeta^2}}} \\
\Delta T &= \frac{\pi}{\omega_0\sqrt{1-\zeta^2}}
\end{aligned}
$$

Figure 4: Three-impulse input — designed to have a vibration-error expression which is both zero and tangent at the expected system natural frequency, ω_0. ζ is the expected damping ratio.

sequences. The sum, however, is an upper bound on the length of the two-mode sequence which can be generated directly by simultaneously solving together the same equations that generated the two individual sequences. For example, if the four equations used to generate the sequence in figure 4 were repeated for a different frequency, a system of eight equations would result and could be solved for four unknown impulse amplitudes and times (plus the first, arbitrary impulse), yielding a five-impulse sequence. The resulting sequence has four fewer impulses than the result of convolving the two independent sequences, and is always shorter in time. An arbitrary number of such sequences can be combined (either by convolution or by direct solution) to generate an input that will not cause vibration in any of the modes that have been included in the derivation.

2.4 Using Impulse Input Sequences to Shape Inputs

Sections 2.2 and 2.3 have presented a method for obtaining a system impulse input sequence which simultaneously eliminates vibration at the natural frequencies of interest and includes robustness to system variability. The impulse sequences shown in figures 2, 4, and 6 are the shortest sequences constructed of only positive impulses which satisfy the constraints in equation 6 and the appropriate derivative constraints. In this sense the sequences are "time-optimal" — no shorter input can be constructed that simultaneously meets the same constraints.

This section presents a method for using the sequences derived above to generate arbitrary inputs with the same vibration-reducing properties. Once the appropriate impulse sequence has been developed, it represents the shortest input that meets the desired design criteria. Therefore, if the system is commanded to make an extremely short move, the best that can be commanded in reality is the multiple-impulse sequence that was generated for the system. Just as the single impulse is the building block from which any arbitrary function can be formed, this impulse sequence can be used as a building block for arbitrary vibration-reducing inputs. The vibration reduction can be accomplished by convolving any arbitrary desired input to the system together with the impulse sequence in order to yield the shortest actual system input that makes the same motion without vibration. The sequence, therefore, becomes a prefilter for any input to be given to the system. The time penalty resulting from prefiltering the input equals the length of the impulse sequence (on the order of one cycle of vibration for the sequences shown in 2.2). Figure 10 shows the convolution of an input (for example, the signal from a joystick in a teleoperated system) with a non-robust, two-impulse sequence.

The impulse sequences from 2.2 have been normalized to sum to one. This normalization guarantees that the convolved motor input never exceeds the maximum value of the commanded input. If the commanded input is completely known in advance for a particular move, the convolved motor input can be rescaled so that the

maximum value of the function is the actuator limit of the system.

The new technique consists of selecting an impulse sequence that has the desired robustness for the system that is to be controlled. The sequence is designed for the natural frequency and damping ratio of the closed-loop system. This input sequence is then convolved with any inputs that are sent to the closed-loop plant. Figure 11 shows a schematic diagram of the implementation of the new technique. Because the input shaping is for the closed-loop system, any controller may be used.

It is important to note that convolution of a physically-realizable requested input and an impulse sequence **always** results in a physically-realizable shaped command input to the system. The convolution process merely superposes time-shifted copies of the original command. Impulses are never sent to the system.

For historical reference, the result of convolving a non-robust two-impulse sequence with a step input yields the Posicast input developed by O.J.M. Smith in 1958 [23]. The robustness plot of figure 3 demonstrates why Posicast is not generally used. For small changes or uncertianties in the system natural frequency, a considerable amount of residual vibration is incurred.

2.5 Application to Nonlinear Systems

No general statement can be made regarding the application of the new technique to nonlinear systems since each nonlinearity poses unique problems. Nonlinearities that tend to appear as shifts in natural frequency do not seem to interfere with the vibration-reducing effects of the new shaping technique because of the robustness to frequency uncertainty that was included in the derivation. Many simulations of geometrically nonliear systems have been performed. An example of such a simulation is provided below. As long as the system is varying slowly, the new shaping technique tends to work (at least on the nonliear, manipulator systems that were considered). A more detailed discussion of nonlinear systems is presented in Singer [22].

2.6 Results

The shaped commands are tested on a computer model of the Space Shuttle Remote Manipulator System (RMS). The computer simulation was developed by Draper Laboratories for use by NASA to verify and to test payload operations. The Draper shuttle model (DRS) includes many of the nonlinear complicating features of the hardware Shuttle manipulator such as stiction/friction in the joints; nonlinear gearbox stiffness; asynchronous communication timing; joint freeplay; saturation; digitization effects; and the nonlinear spacial frequency shifts of the three-dimensional RMS. The simulation was verified with actual space-shuttle flight data. Excellent agreement was obtained both for steady-state and for transient behavior.

Figure 12 shows the response of the DRS to a 4 second pulse velocity command

from the astronaut operator and response to the same command shaped by the three-impulse sequence described above. The residual vibration for this move is reduced by (a factor of 25for the unloaded shuttle arm. Comparable results were obtained for a variety of moves tested. The fact that this simulation model is highly nonlinear demonstrates that this method can work even in the presence of certain system nonlinearities.

3 Conclusion

The use of shaped inputs for commanding computer-controlled machines shows that significant vibration reduction can be achieved. The cost in extended move time is small (on the order of one cycle of vibration), especially when compared to the time saved in waiting for settling of the machine's vibrations. A straightforward design approach for implementing this preshaping technique has been presented along with some results from Draper Laboratory's Space Shuttle manipulator model.

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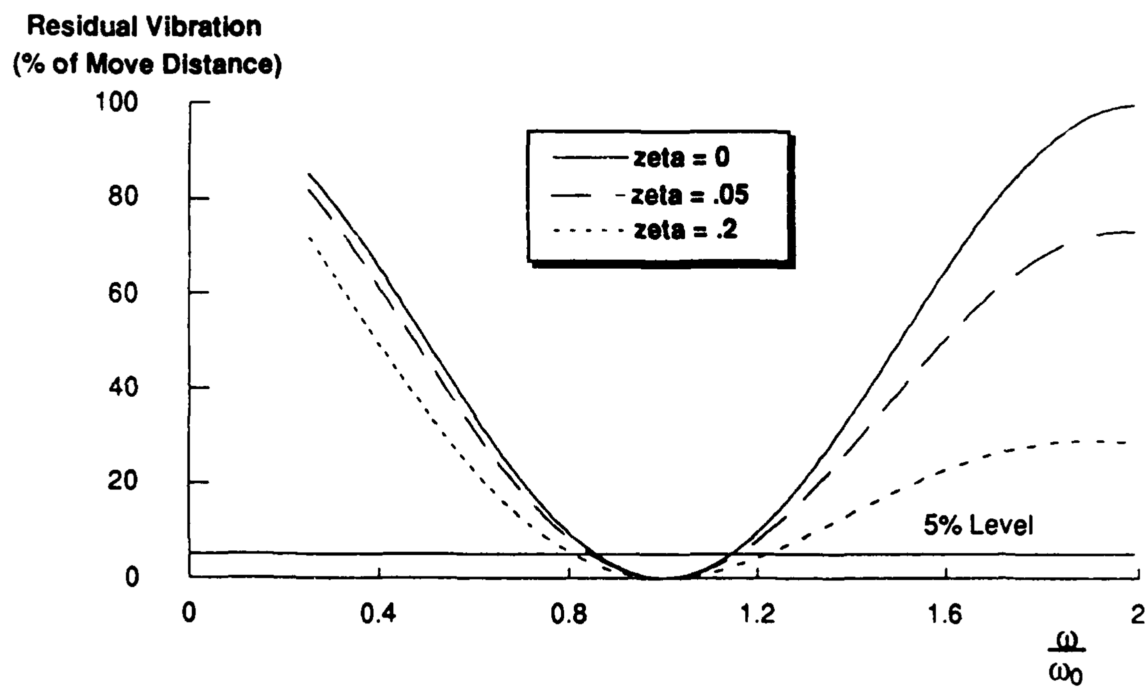
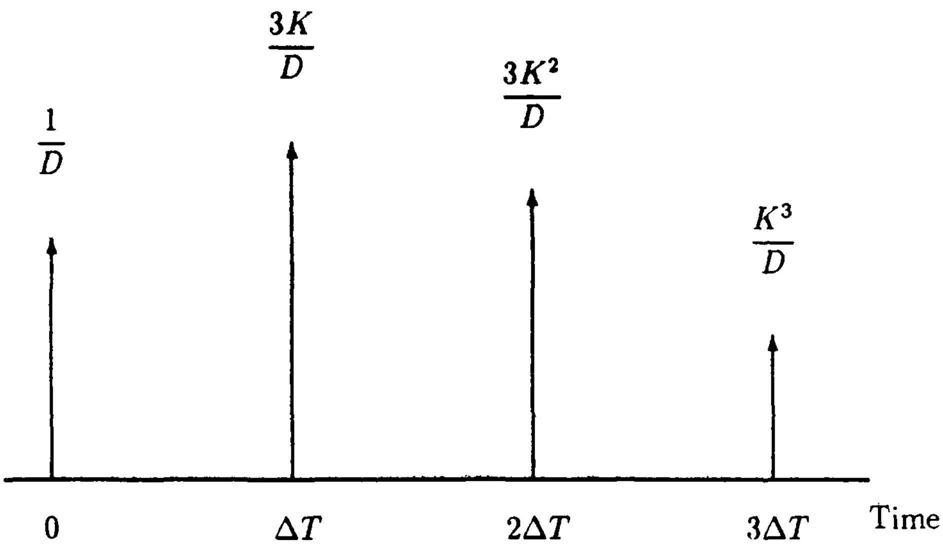


Figure 5: Vibration error vs. system natural frequency for three systems with different values of damping ratio excited by the three-impulse sequence in figure 4.



$$
\begin{aligned}
K &= e^{-\frac{\zeta\pi}{\sqrt{1-\zeta^2}}} \\
\Delta T &= \frac{\pi}{\omega_0\sqrt{1-\zeta^2}} \\
D &= 1 + 3K + 3K^2 + K^3
\end{aligned}
$$

Figure 6: Four-impulse input — designed to have a vibration-error expression which is zero, tangent, and flat at the expected system natural frequency, ω_0. ζ is the expected damping ratio.

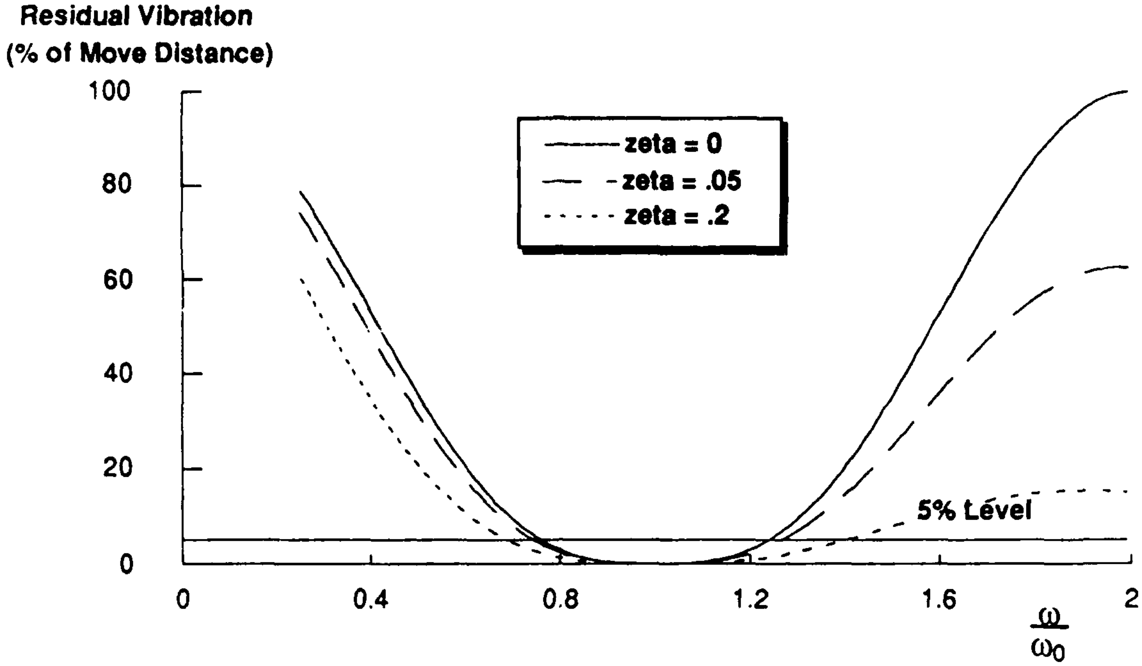


Figure 7: Vibration error vs. system natural frequency for three systems with different values of damping ratio excited by the four-impulse sequence in figure 6.

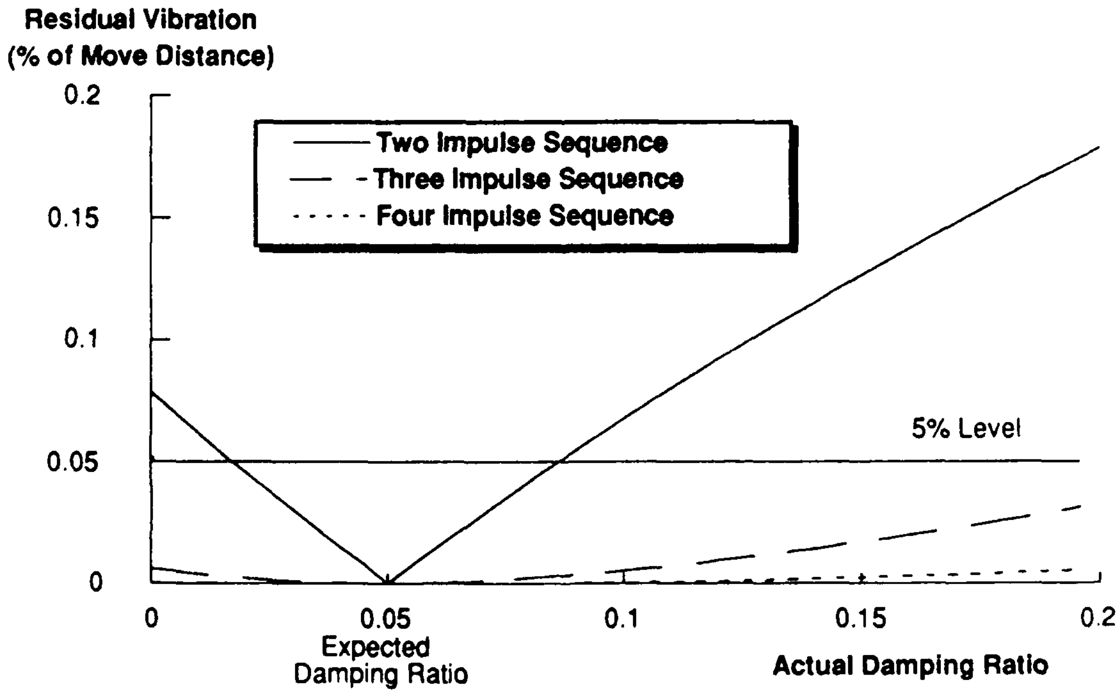


Figure 8: Vibration error vs. damping ratio for the two-, three-, and four- impulse inputs presented in section 2.2.1 calculated for a system with a damping ratio of .05.

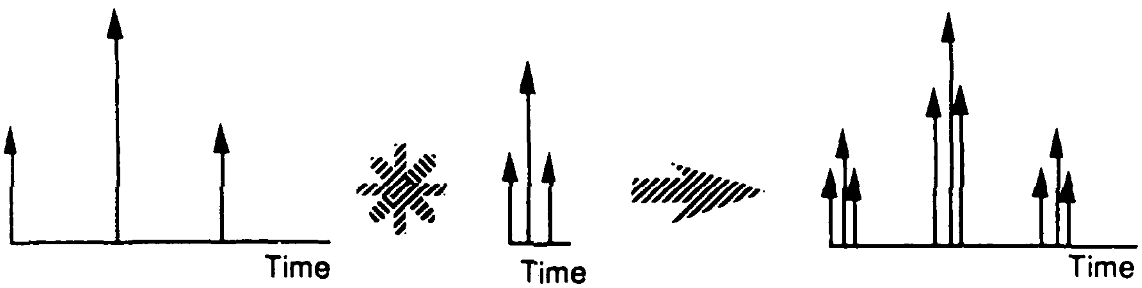


Figure 9: Vibration reduction for several modes. An example of convolving two three-impulse sequences together to form a single sequence that reduces vibration in two separate modes.

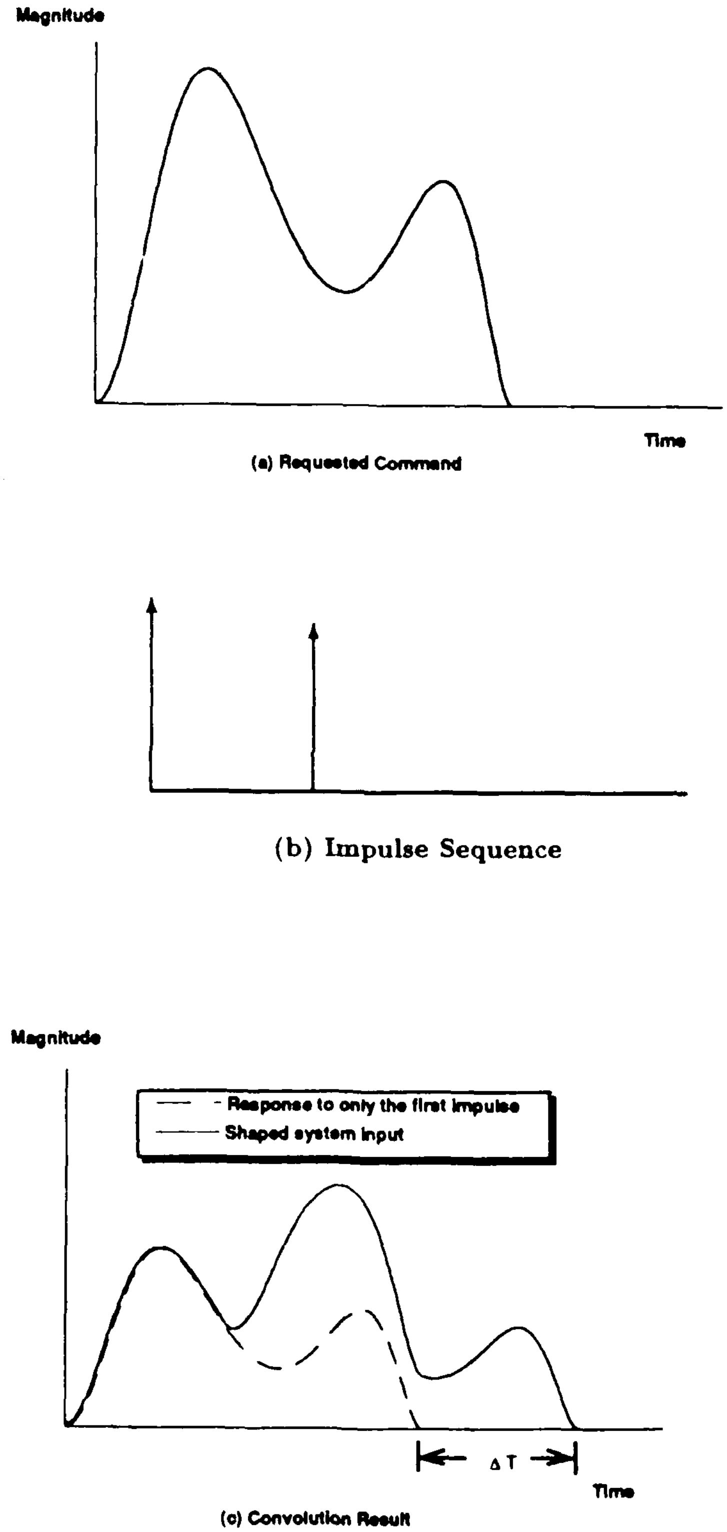


Figure 10: Convolution of a command in (a) with a Two-Impulse Sequence shown in (b) yields the system input shown in (c).

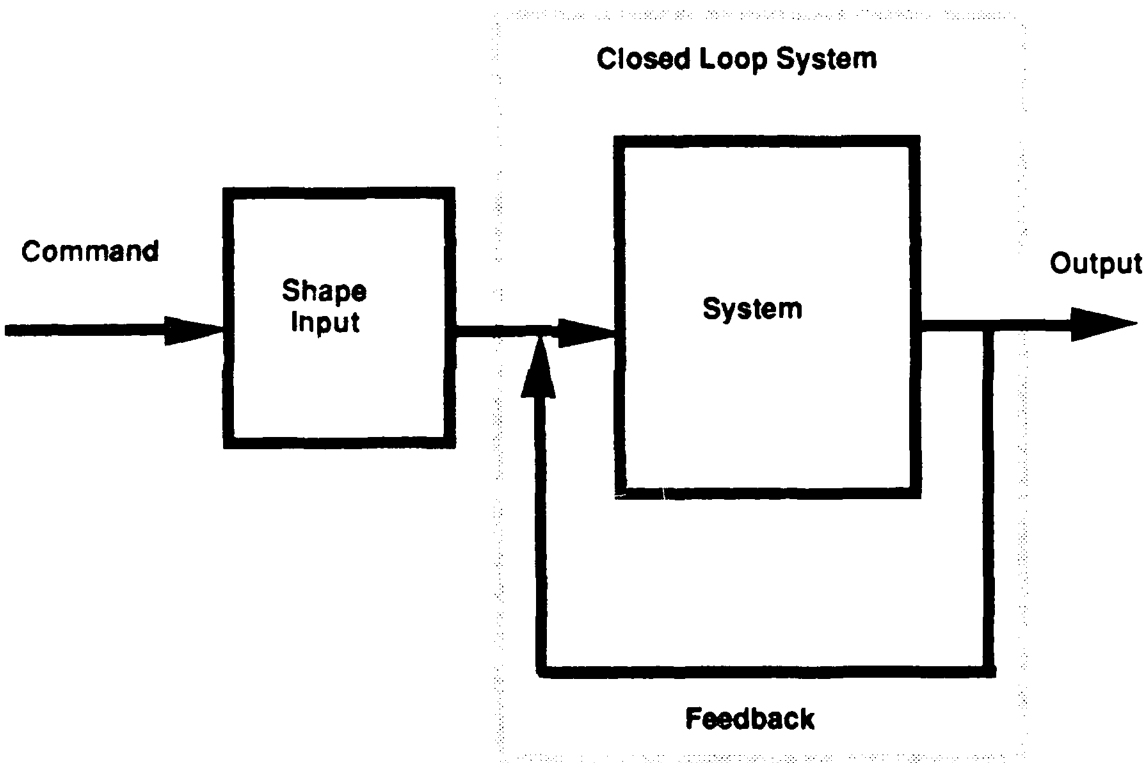


Figure 11: Schematic diagram for implementation of the new technique

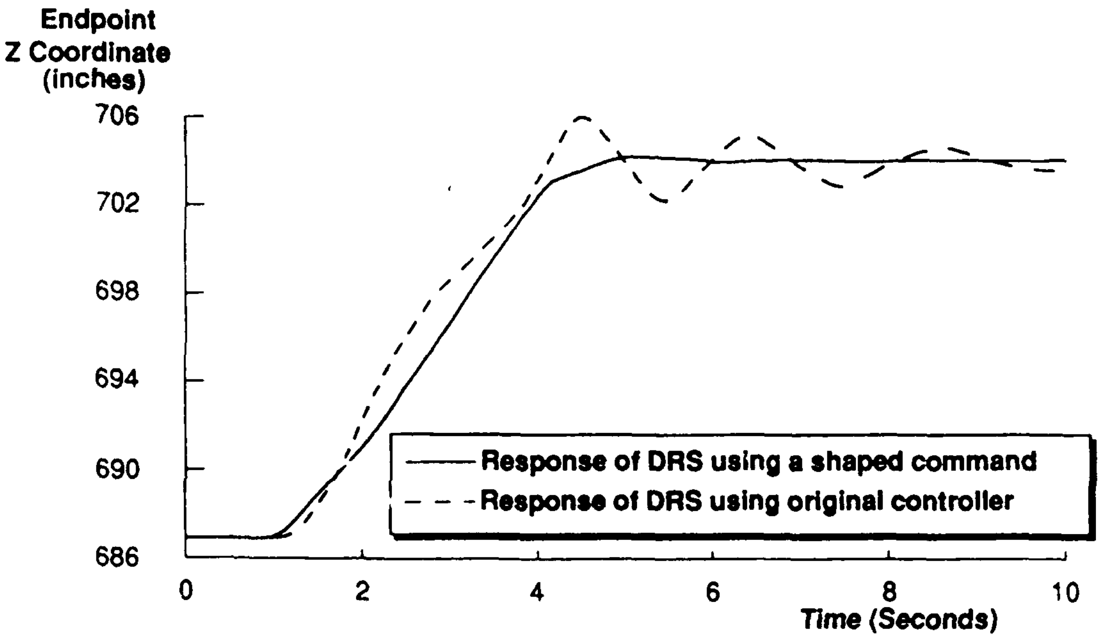


Figure 12: Comparison between the response of the DRS using the original RMS controller (shown as dotted) and the response generated by shaping the same command with a three-impulse sequence (shown as solid). The reduced vibration of the solid curve over the dotted curve is a direct result of preshaping of the input command.